AD-A094 752

AD

AD-E400 464

Contractor Report ARLCD-CR-80030

DESIGN OF WASTE HEAT BOILER FOR SCRANTON ARMY AMMUNITION PLANT

Robert J. Krowech
MECHANICAL TECHNOLOGY INCORPORATED
Latham, New York

George Scullin, Project Engineer
ARRADCOM
Dover, New Jersey

August 1980

US ARMY ARMAMENT RESEARCH AND DEVELOPMENT COMMAND
LARGE CALIBER
WEAPON SYSTEMS LABORATORY
DOVER, NEW JERSEY

SECURITY CLASSIFICATION OF THIS PAGE *(When Data Entered)*

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER
Contract Report ARLCD-CR-80030

2. GOVT ACCESSION NO.

3. RECIPIENT'S CATALOG NUMBER

4. TITLE *(and Subtitle)*
Design of Waste Heat Boiler for
Scranton Army Ammunition Plant

5. TYPE OF REPORT & PERIOD COVERED
Final

6. PERFORMING ORG. REPORT NUMBER
80TR64

7. AUTHOR(s)
Robert J. Krowech, Mechanical Technology Incorporated
George Scullin, Project Engineer, ARRADCOM

8. CONTRACT OR GRANT NUMBER(s)
DAAK10-79-C-0406

9. PERFORMING ORGANIZATION NAME AND ADDRESS
Mechanical Technology Incorporated
968 Albany-Shaker Road
Latham, New York 12110

10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS

11. CONTROLLING OFFICE NAME AND ADDRESS
ARRADCOM, TSD
ATINFO (DRDAR-TSS)
Dover, NJ 07801

12. REPORT DATE
August 1980

13. NUMBER OF PAGES
51

14. MONITORING AGENCY NAME & ADDRESS *(if different from Controlling Office)*
ARRADCOM, LCWSL
Munitions Systems Division (DRDAR-LCU-M)
Dover, NJ 07801

15. SECURITY CLASS. *(of this report)*
Unclassified

15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT *(of this Report)*
Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT *(of the abstract entered in Block 20, if different from Report)*

18. SUPPLEMENTARY NOTES

19. KEY WORDS *(Continue on reverse side if necessary and identify by block number)*
Waste heat recovery
Boiler
Steam

20. ABSTRACT *(Continue on reverse side if necessary and identify by block number)*
The Army Ammunition Plant at Scranton, Pennsylvania, has rotary hearth furnaces to heat billets for the forging operation. Mechanical Technology Incorporated established the economic and operational feasibility of waste heat recovery from the most used forged furnaces. The flue gas flow and temperature were measured, and an energy balance was performed. Gas flow and temperature were found suitable for a small waste heat boiler. During the course of testing a flow control problem was discovered. MTI suggests that this defect be corrected before the installation of a waste heat recovery system. After exploring several concepts, MTI designed a

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE *(When Data Entered)*

waste heat recovery system having minimal influence on the forging operation. The payback period of 1.54 years is attractive. Also, a rigorous economic evaluation (the present worth analysis), indicates that a waste heat recovery system would be a valuable addition to the Scranton plant.

TABLE OF CONTENTS

Page No.

TABLES

FIGURES

INTRODUCTION

In response to the award of contract number DAAK10-79-C-0406, Mechanical Technology Incorporated (MTI) conducted a study for the "Design of a Waste Heat Boiler for the Scranton Army Ammunition Plant (SAAP)," Scranton, Pennsylvania. The purpose of the study was to assess the economic feasibility of heat recovery from furnace flue gas, and to conceptually design and specify a heat recovery system.

The study was completed in three phases to allow for a broad-scoped investigation. The first phase defined the requirements for the heat recovery system, while the second phase explored four system concepts and selected the most economically attractive system. The third phase developed the design and specifications of the system.

MTI's background and expertise in waste heat recovery, economic analysis and hardware costing provided a strong base of knowledge for the duration of this study. Also, MTI's familiarity with the SAAP installation (based on earlier work in preparing conceptual system representations for a heat recovery/electrical power-generating unit utilizing the plant's stack gas), was effectively used in the ultimate specification of a waste heat recovery boiler. Hence, an economically attractive system, well suited to the capabilities and intended service at the plant, resulted.

Because the exhaust gases from the billet-heating furnaces at SAAP represent a potential source of waste energy, this study takes on added significance in that it meets one of the primary objectives of the current

national energy situation; namely, the extraction of energy from a fuel-fired operation. Most importantly, it reaffirms the application of waste-generated energy to produce plant steam, and, additionally, it confirms this energy conversion approach as a viable and cost-effective alternative.

Operation of Forge Furnaces at SAAP

The two SAAP furnaces selected for heat recovery are used for manufacturing the 155mm M107 projectile. Under current production, the furnaces are used one at a time at two-week intervals; this places a major constraint on the design of the waste heat boiler system.

Because the furnace operating cycle determines the amount of heat available for steam production and the economic potential for waste heat recovery, a breakdown of the operating conditions of the furnace is provided as follows:

1. The furnace operates in production two shifts per work day for five days a week.
2. The furnace idles (loaded with billets) on the third shift for four days a week and is shut down on Friday night.
3. The furnace is shut down for 27 hours during the weekend.
4. The furnace is restarted during the weekend for a period of 29 hours.

In order to establish the background for the stack and damper design considerations, the basic operation of the furnace is outlined here. The forge furnaces at SAAP are rotary hearth furnaces. Each furnace con-

tains a total of 61 rows with each row three billets deep, with a cycle time of one hour. Billets are simultaneously loaded and unloaded at a rate of three per minute and the furnace hearth is indexed once every minute or 60 times per hour. The furnace has four firing zones with air to natural gas ratios of 10:1, 10:1, 6:1, 6:1 for zones 1, 2, 3 and 4, respectively. The temperature of zones 1 and 2 is 1800°F; zone 3 is 2000°F and zone 4 is 2200°F, thus heating the billets to approximately 2000°F in one hour. The purpose of zones 3 and 4 having non-stoichiometric ratios is to prevent oxidation of the steel as it reaches the higher temperatures experienced in these zones. The excess natural gas resulting from the non-stoichiometric conditions in zones 3 and 4 is combusted outside of the furnace in the section of the stack where the measurements are taken. At this point, completion air is provided to combust the natural gas and dilution air is added to reduce the temperature of the flue gas so the recuperator is not overheated. Finally, a damper is provided in the stack to create a positive pressure in the furnace of approximately 0.5 in. water; this minimizes air from entering the furnace and oxidizing the billets.

FIELD SURVEY OF FURNACE WASTE HEAT OPPORTUNITIES

In order to define the requirements for the waste heat recovery system, the study began with a field survey that sought to determine stack gas characteristics necessary for boiler design criteria. Gas stream measurements of the current furnace system, taken in both the run mode and the idle mode, were used to determine the flow quantity and temperature that would be available to a waste heat boiler. In addition, these measurements provided the basis for a furnace energy balance, as well as an assessment of loading door losses.

Run Mode Testing

The flue gases were measured at the point indicated in figures 1 and 2; this point was chosen because the preceding straight section tended to make the flow somewhat uniform. Since the velocity profile was expected to be symmetrical from left to right, a vertical traverse was selected.

In order to calculate velocity measurements, the flow (stagnation pressure) was measured with a pitot tube and a slant manometer. The 1.245 m (4'1") inside diameter (ID) was traversed twice, with measurements taken at 0.152 m (6") intervals. Example calculations are shown in appendix A.

Temperature measurements were taken with a type-K thermocouple and a digital readout. The traversing and averaging process were the same as for the velocity measurements.

Furnace skin temperatures were taken at a later date with the furnace operating under a similar loading; errors introduced by the taking of these measurements at another time were very small. As the skin heat loss varied only slightly over the range of operating conditions, these losses represented only a small fraction of the total energy. From the skin temperature, the heat loss was calculated using common heat loss charts.

The energy balance for the furnace while heating billets is shown in table 1. Figure 3 graphically shows the energy distribution.

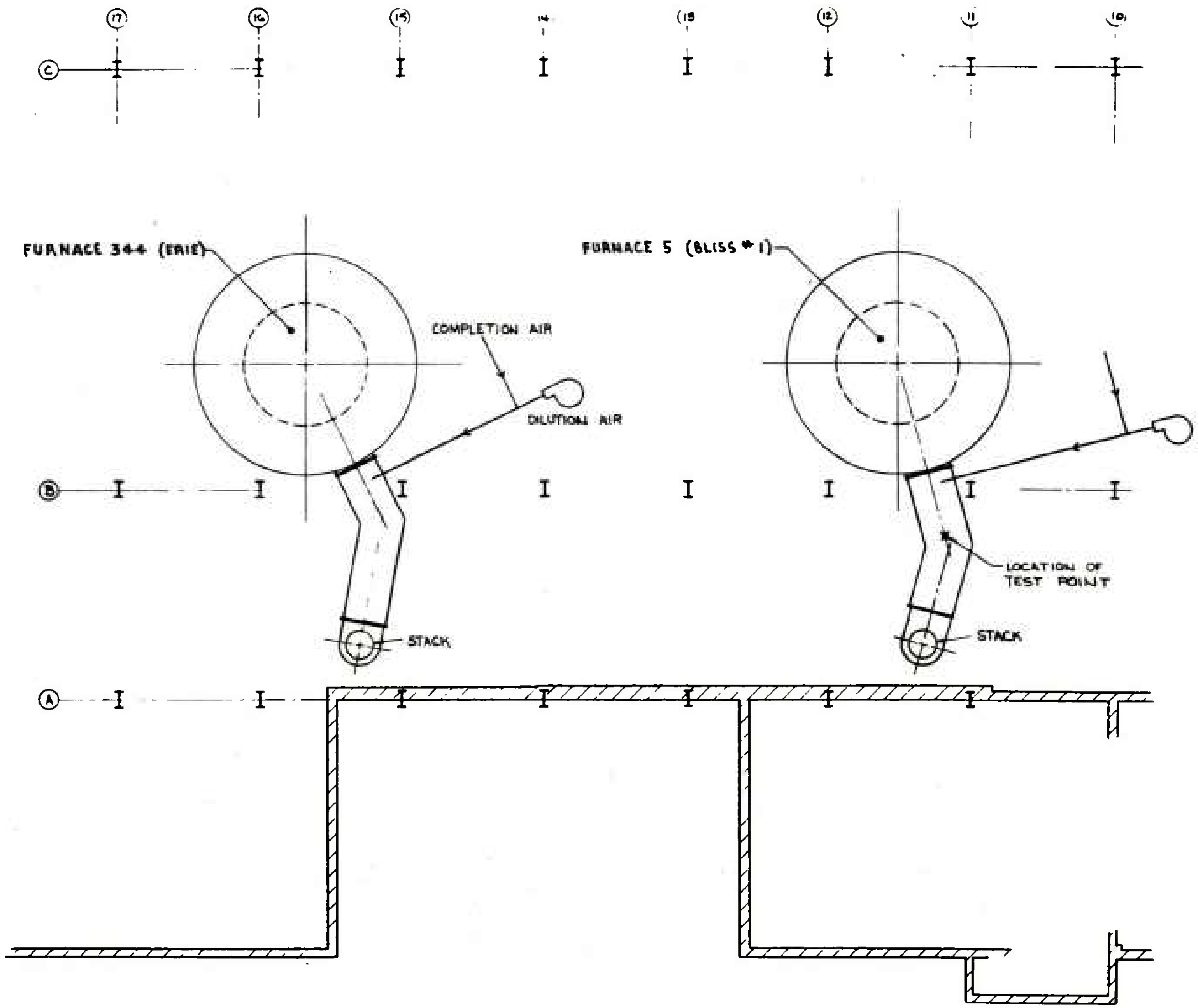

Figure 1. Plan view of existing system

802014

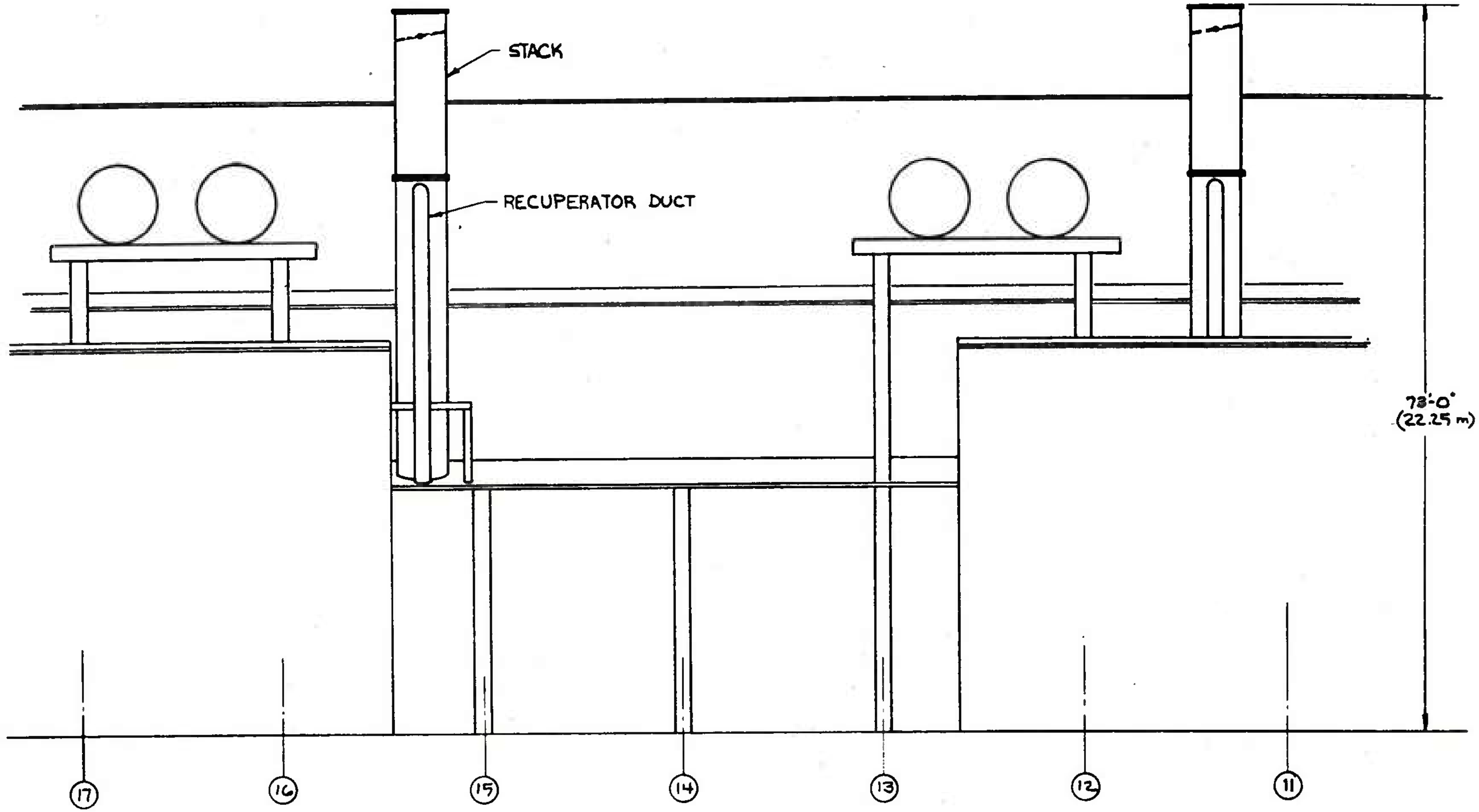

Figure 2. Elevation of existing system

802013

Table 1. Energy balance - furnace running billets (as measured)

Item	Flow	Temperature °C(°F)	Energy kcal x 10^6 (Btu/hr x 10^6)	Comments
Natural Gas	5935 m^3/h (20,953 scf/hr)		4.89 (19.40)	Based on LHV
Recuperative Air	2722 kg/h (6000 lb/hr)	437 (820)	0.28 (1.11)	
TOTAL ENERGY IN			5.17 (20.51)	
Flue	10,224 kg/h (22,540 lb/hr)	1025 (1877)	2.96 (11.73)	
Billets	180/hr	1093 (2000)	1.58 (6.26)	
Skin		121-204(250-400)	0.35 (1.38)	
Door Radiation		1777,1055(2150,1950)	0.10 (0.41)	
Door Flow	948 kg/h (2090 lb/hr)	1110,1082(2030,1980)	0.28 (1.10)	Approximate
TOTAL ENERGY OUT	948 kg/h (2090 lb/hr)		5.27 (20.88)	
Error			0.10 (0.37)	

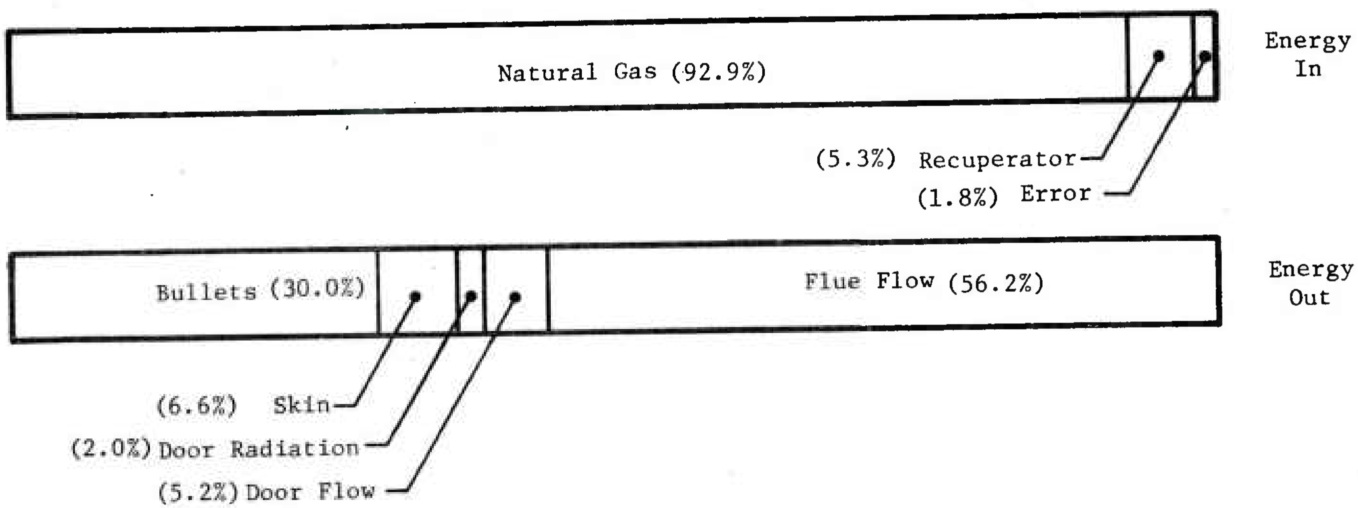

Figure 3. Energy bar chart

802017

Idle Mode Testing

The measurements with the furnace in the idle mode were taken after the furnace had idled at least 45 minutes to come to steady-state operation; however, because the dilution air automatic control was determined to be hunting, no steady-state flow or temperature measurements were attainable. Instead, two sets of measurements were taken: one at the maximum flow, and another at the minimum flow.

In testing, the period of oscillation was 2 min 40 sec, while the flow ranged from 5,475 kg/h (12,070 lb/hr) to 29,520 kg/h (65,080 lb/hr). This oscillation, as observed in the idle mode, is undesirable for a heat recovery system or for any system; it is caused by several different factors. One possibility is the nonlinearity of the dilution air valve itself. At the reduced flow of idling, the characteristics of the valve are changed, but the characteristics of the controller are not. Another possibility is that during idling, when the fuel-rich zones are being fired, small amounts of dilution air are acting like completion air. This small increase in air flow raises the flue gas temperature. After stoichiometeric ratio is reached, the process reverses and valve makes the proper temperature correction. However, the temperature overshoots the set point each time.

The energy loss (skin loss, etc.) was taken to be the same as it was during the run mode. The use of natural gas was measured for the duration of the test, and the recuperator air flow was steady and measured. Since the energy entering the system was accurately known, the average flue gas flow was calculated based on an assumed average temperature. The average

temperature was chosen at 1149°C (2100°F), because the actual energy flux was closer to the low-flow condition. The mass flow was calculated again, based on what it would be while maintaining a steady 927°C (1700°F) set point. The values are shown in table 2.

Energy Losses from Loading Doors

Among the major concerns of the study was the significance of the energy lost from the loading doors. The results of this investigation can be summarized as follows.

Energy is lost out of the loading doors in two ways: through radiation heat from the furnace interior to the surrounding area, and through the outward flow of hot gas. While the radiation loss was straightforward (any error would be small), the flow loss was complex, as it interacted with continuing combustion, entrained outside air, and natural convection. Therefore, to establish an order of magnitude, crude measurements of velocity and temperature were taken. Because all other energy flows were known to reasonable accuracy, the flow losses from the loading doors were established from the energy balance. The crude estimate also served as a check should other quantities be in error.

The energy lost from the loading is approximately 0.103×10^6 kcal/h (0.41×10^6 Btu/hr) via radiation, and roughly 0.277×10^6 kcal/h (1.1×10^6 Btu/hr) via outward flow. The total lost energy is roughly 0.378×10^6 kcal/h (1.5×10^6 Btu/hr) or about 8% of the energy available from the fuel.

Table 2. Energy balance - idling

Source	Flow	Temperature °C(°F)	Energy kcal x 10^6 (Btu/hr x 10^6)	Comments
Natural Gas	378 m^3/h (13,348 scf/hr)		3.12 (12.36)	Using LHV
Recuperative Air	1887 kg/h (4160 lb/hr)	443 (830)	0.20 (0.78)	
TOTAL ENERGY IN			3.32 (13.14)	
Flue:				
High	29,515 kg/h (65,080 lb/hr)	893 (1640)	6.93 (27.49)	
Low	5,474 kg/h (12,070 lb/hr)	1329 (2424)	2.16 (8.59)	
Avg.	8,268 kg/h (18,230 lb/hr)	1149 (2100)	2.69 (10.66)	Calculated based on 1149°C and 927°C (2100°F and 1700°F)
Skin		121-204 (250-400)	0.35 (1.38)	
Door		1066 (1950)	0.04 (0.17)	Radiation
	830 kg/h (1830 lb/hr)	971 (1780)	0.23 (0.93)	Flow

Some heat recovery is possible if the flow from the doors is ducted to the waste heat boiler. Such a system would have hoods as near as possible to the doors, insulated ducts to the boiler inlet, and dampers to shut off the flow when the doors are closed.

Not all of the 0.277×10^6 kcal/h (1.1×10^6 Btu/hr) energy loss can be recovered. Suppose, for example, that a carefully placed hood collected 75% of the escaping flow and diluted it with an equal quantity of room air. The resulting temperature would be approximately:

$$\frac{1093°C + 21°C}{2} = 557°C \left(\frac{2000°F + 70°F}{2} = 1035°F.\right)$$

Allowing for some loss in the duct, the flow arriving at the boiler would be about 538°C (1000°F). The specific heat in this range is 0.26 cal/g°C (Btu/lb°F) so the heat recovery would be:

$$0.26 \times (1000 - 250) \times 2090 \times 0.75 \times 2$$
$$= 0.15 \times 10^6 \text{ kcal/h } (0.61 \times 10^6 \text{ Btu/hr}).$$

This represents an increase of 272 kg/h (600 lb/hr) of steam production.

During the heating season, the flow escaping from the loading doors contributes to the heating of the forge shop. The above example shows that 0.277×10^6 kcal/h (1.1×10^6 Btu/hr) worth of heating is replaced by 0.15×10^6 kcal/h (0.61×10^6 Btu/hr). Additionally, withdrawing air from the building requires that it be replaced with outdoor air. Using a specific heat of 0.24 and an outdoor temperature of -18°C (0°F), this additional loss is:

$$0.24 \times (70 - 0) \times 2090 \times 0.75 \times 2$$
$$= 0.01 \times 10^6 \text{ kcal/h } (0.05 \times 10^6 \text{ Btu/hr}).$$

Thus, during part of the heating season, heat recovery from the loading doors is, in fact, a heat loss.

On a annual basis, a well-designed system recovering heat from the loading doors can make a small contribution toward energy costs. A system not carefully designed and controlled can have the opposite effect.

Efficiency of Waste Heat Boilers

Boiler efficiency, an important and meaningful concept in relation to fired boilers, is defined as follows:

$$\text{Efficiency} = \frac{\text{Steam energy}}{\text{Fuel energy}}$$

Developing a parallel efficiency for waste heat boilers, the concept becomes:

$$\text{Efficiency} = \frac{\text{Steam energy}}{\text{Energy available}}$$

The energy available is the sensible heat plus the latent heat of the flue gas or

$$\left[\bar{C}_{p1}\ (T_1 - T_{amb}) + Xh_{f-g}\right]\dot{m} = \text{Energy available}$$

Where:

$\bar{C}_{p1}$ = Average flue gas specific heat from T_1 to T_{amb}

T_1 = Initial (high) flue gas temperature

T_{amb} = Ambient temperature, which is the lowest temperature to which the flue gases can be cooled

h_{fg} = Latent heat of the water vapor in kcal/kg (Btu/lb).

$\dot{m}$ = Mass flow of the flue gas

Note that latent heat is almost never from flue gas, but is traditionally included in the available energy.

The net energy absorbed by the steam will be equal to the energy given by the flue gas less any heat losses through the casing or:

$$\text{Steam energy} = \bar{C}_{p2} \ (T_1 - T_2) - Q \text{ casing}$$

Where:

$\bar{C}_{p2}$ = Average flue gas specific heat from T_1 to T_2

T_2 = Temperature of flue gas exiting the heat recovery system

Q casing = Casing heat loss

Looking at the expression for Energy available, it is obvious that T_{amb} is rather arbitrary and must be specified if efficiency is to be well defined. The other terms are not arbitrary, but must be accurately known if efficiency is to be specified.

The boiler manufacturer controls two quantities in the expression for Steam energy, T_2 and Q casing. Thus, specifying the above two quantities is equivalent to specifying boiler efficiency, but with the advantage of not having to select T_{amb} or accurately know the latent heat in the flue gas.

T_2 has been chosen at 121°C (250°F). As will be explained in the Economic Analysis section, this selection maximizes heat recovery without requiring excessive heating surface.

Error in Measured Values

The measured heat flows show 0.094 x 10^6 kcal/h (0.37 x 10^6 Btu/hr) less energy entering the control volume than leaving it (a 2% error based on the entering energy). However, for purposes of specifying a waste heat boiler, the flows and temperatures as measured are adequate. These measured values are also adequate for purposes of the economic analysis. Nonetheless, this error is worth some discussion and need not be a mystery.

Because the flow out the doors is complicated (with combined natural convection and forced flow, combustion external to the furnace, and entrained outside air), the measurement of outward flow is only a crude approximation. The real flow out the doors should fall in the range of ½ to 2 times the measured value. This flow should be determinable from the energy balance. Thus, a strong possibility exists that the entire error is attributable to the doors.

The other possibility relates to the measured flue gas flow. Error analysis shows that the energy flow should be within 10% of the measured value. The maximum combined temperature and flow error could be about 0.298 x 10^6 kcal/h (1.17 x 10^6 Btu/hr).

Each of the measurements is subject to some error. The two quantities just discussed (the flue gas energy flow and the door energy flow) are the least accurate as measured, and a detailed analysis of the other quantities would serve no useful purpose. In subsequent work, the most conservative values will be used in each case.

DEVELOPMENT OF SYSTEM CONCEPTS

Based on the field survey results, several concepts were considered for a waste heat recovery system. A discussion of each follows:

Concept 1

The boiler is on the roof of the Forge Shop; the existing stack (recuperator) is replaced with a tube-type recuperator (figure 4).

Advantages are:

1. Ducting is minimized, yielding a clean compact system.
2. Higher preheat temperatures can be attained with a tube-type recuperator.

Disadvantages are:

1. The present system has a delicate control scheme although an ID fan with a control damper could duplicate the present system when the boiler is on-line. However, when no need exists for steam or when the boiler is down for maintenence, a control system much like the present one would be required.

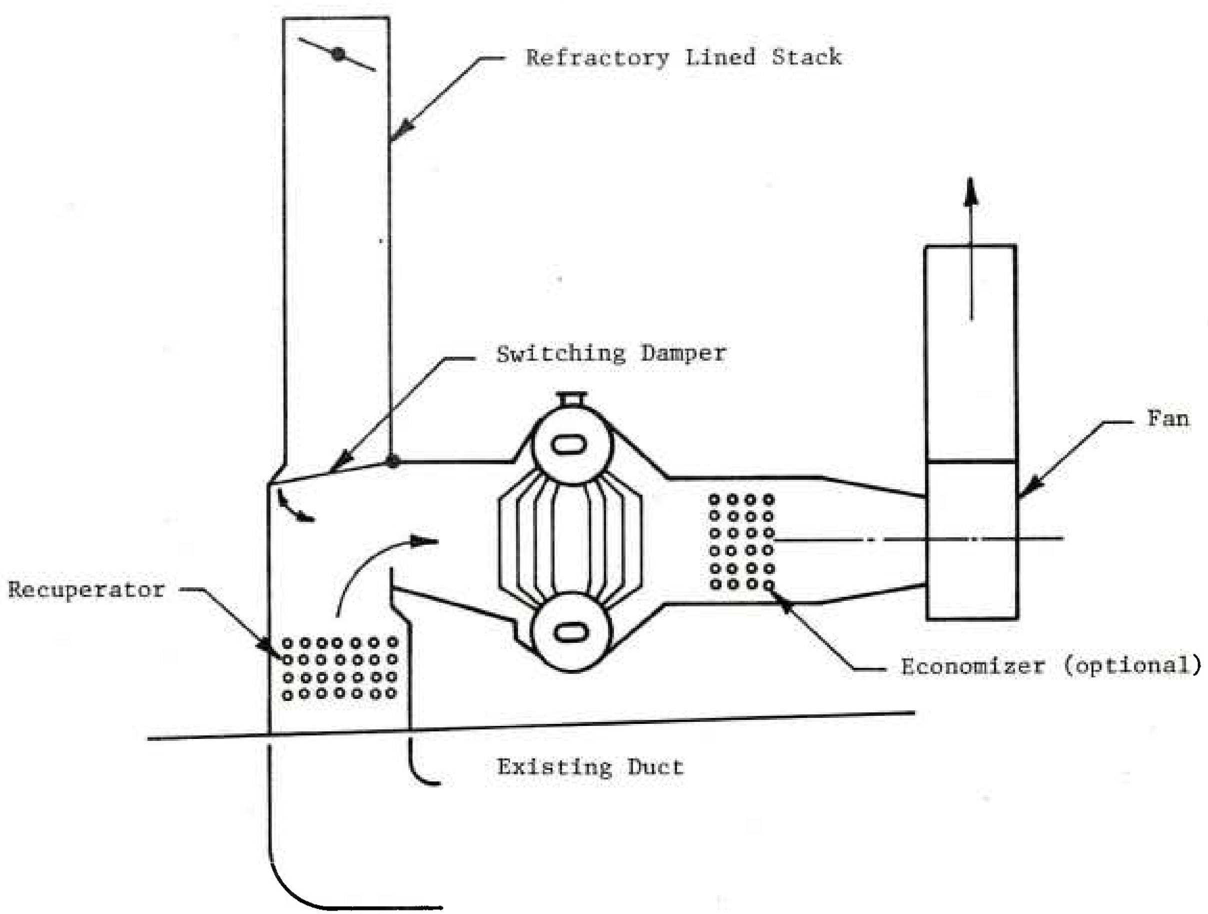

Figure 4. Concept 1

803226

2. The present preheat temperatures are just within the acceptable range for carbon steel, which leads to the belief that carbon steel is used between the recuperator and the furnace. Any significant increase in recuperative preheating, therefore, would require a change to a heat-resistant alloy.
3. Recuperators are very expensive because of the super alloys required.
4. This concept is closer to a system redesign rather than a simple retrofit. Replacement of the recuperator, not including modifications to the stack, would cost approximately $50,000.

Concept 2

The boiler is on the roof, ducted from the top of the existing stack, with one boiler per exhaust stack.

Advantages are:

1. Because the existing recuperator and stack are left unaltered, the furnace can be operated as it is presently whenever the boiler is off-line.

Disadvantages are:

1. Two boilers would be required. Each boiler would operate only half of the time, thus resulting in higher operating costs to produce steam from the total energy generated during current production.

2. Additional ducting at a cost of $10,000 would be required.

Concept 3

The boiler is sized to handle the capacity of one furnace, but ducted to two furnaces.

Advantages are:

1. The exhaust from either of two furnaces can be used to generate steam without duplication of the boiler, fan, stack, cold-weather protection, and much of the additional cost associated with this hardware.
2. The fan is sized for the normal flow rate and does not unnecessarily use up electrical power.

Disadvantages are:

1. The boiler cannot generate twice the steam if two furnaces are used at once (mobilization).
2. If the boiler is down for repair, no waste heat steam can be generated.
3. Additional ducting is required.

Concept 4

One boiler, ducted to two stacks, is sized to handle the capacity of both furnaces.

Advantages are:

1. The boiler can generate steam from both furnaces at once (mobilization).
2. Compared to concept 2, much duplication of equipment is eliminated.
3. When just one furnace is in use, the large boiler generates slightly more steam than would a small one.

Disadvantages are:

1. Two fans, or a single oversized fan, are required.
2. Extra ducting is required.

Figure 5 shows a view of system concept 3, while figure 6 illustrates concept 4. Figure 7 is a flow schematic of the conceptualized system.

Concepts 1 and 2 were rejected because they clearly have poorer economic and operational performance characteristics, as compared to concepts 3 and 4. Because of their similarity in economic advantage, concepts 3 and 4 were selected as the best potential candidates for the waste heat recovery system; hence, an economic analysis was undertaken.

ECONOMICS OF WASTE HEAT-GENERATED STEAM

The economic analysis included an investigation of the potential value of the steam generated by the waste heat boiler. This value is an important consideration as any steam generated beyond what is immediately needed is worthless.

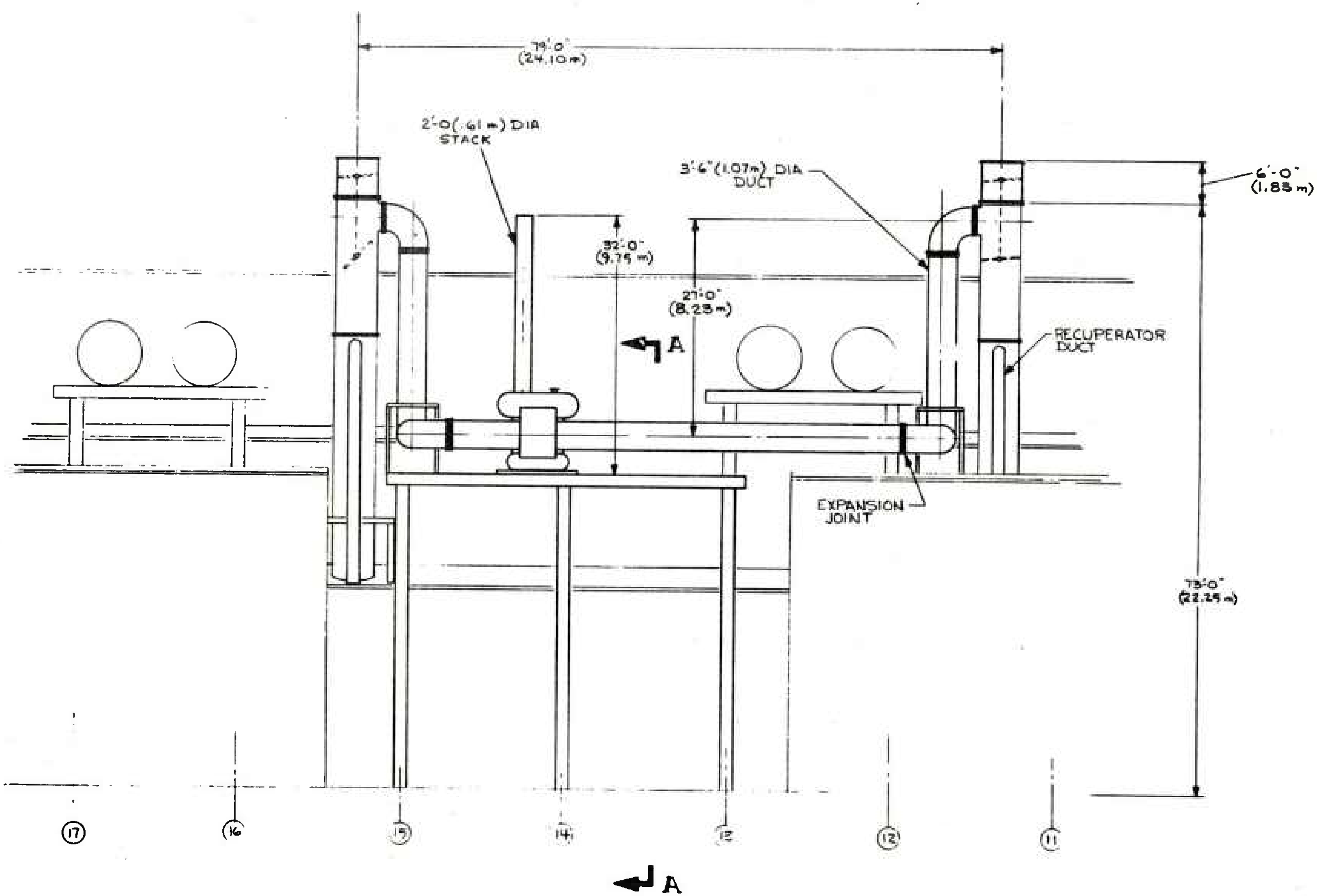

Figure 5. Heat recovery, concept 3

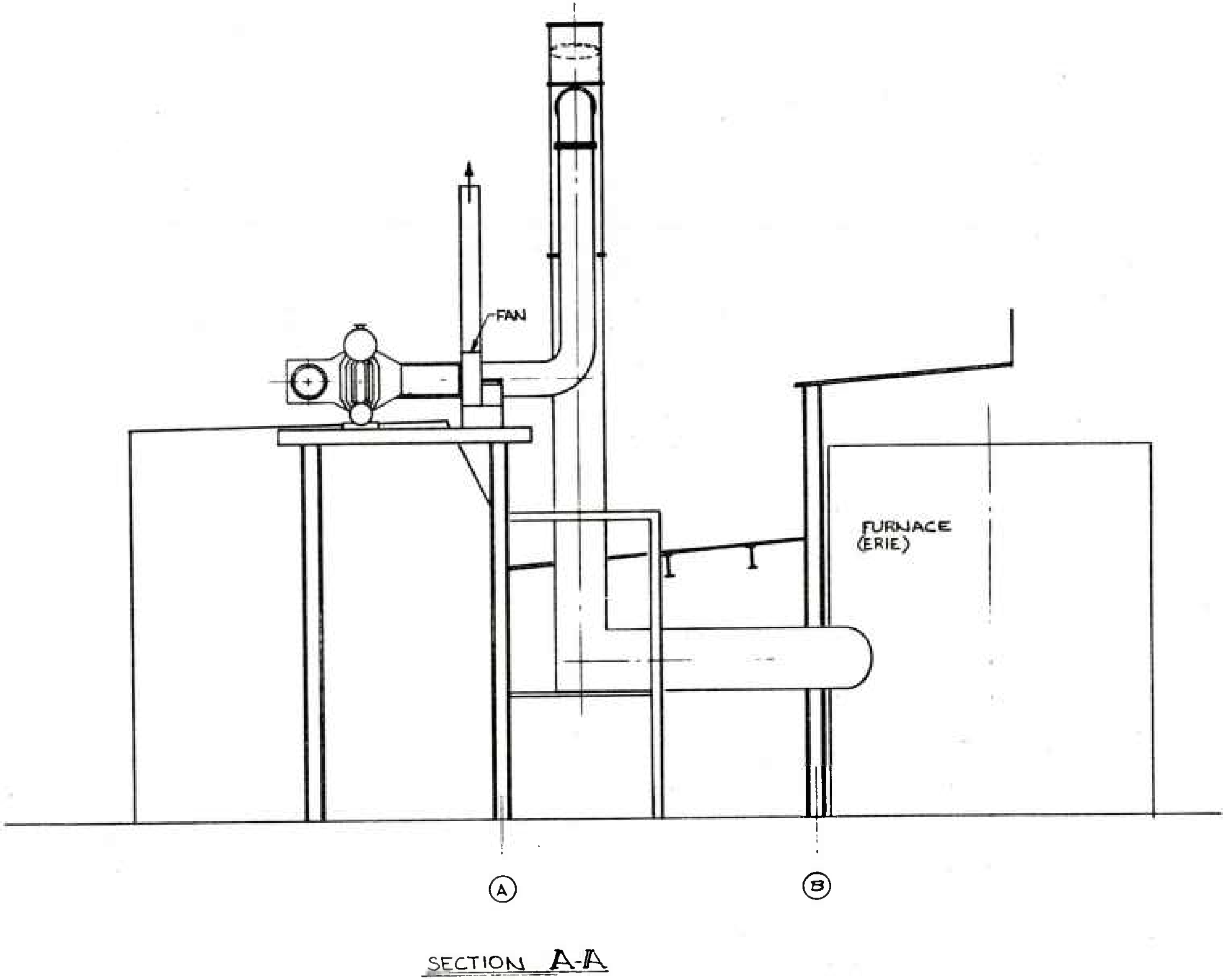

Figure 6. Heat recovery, concept 4

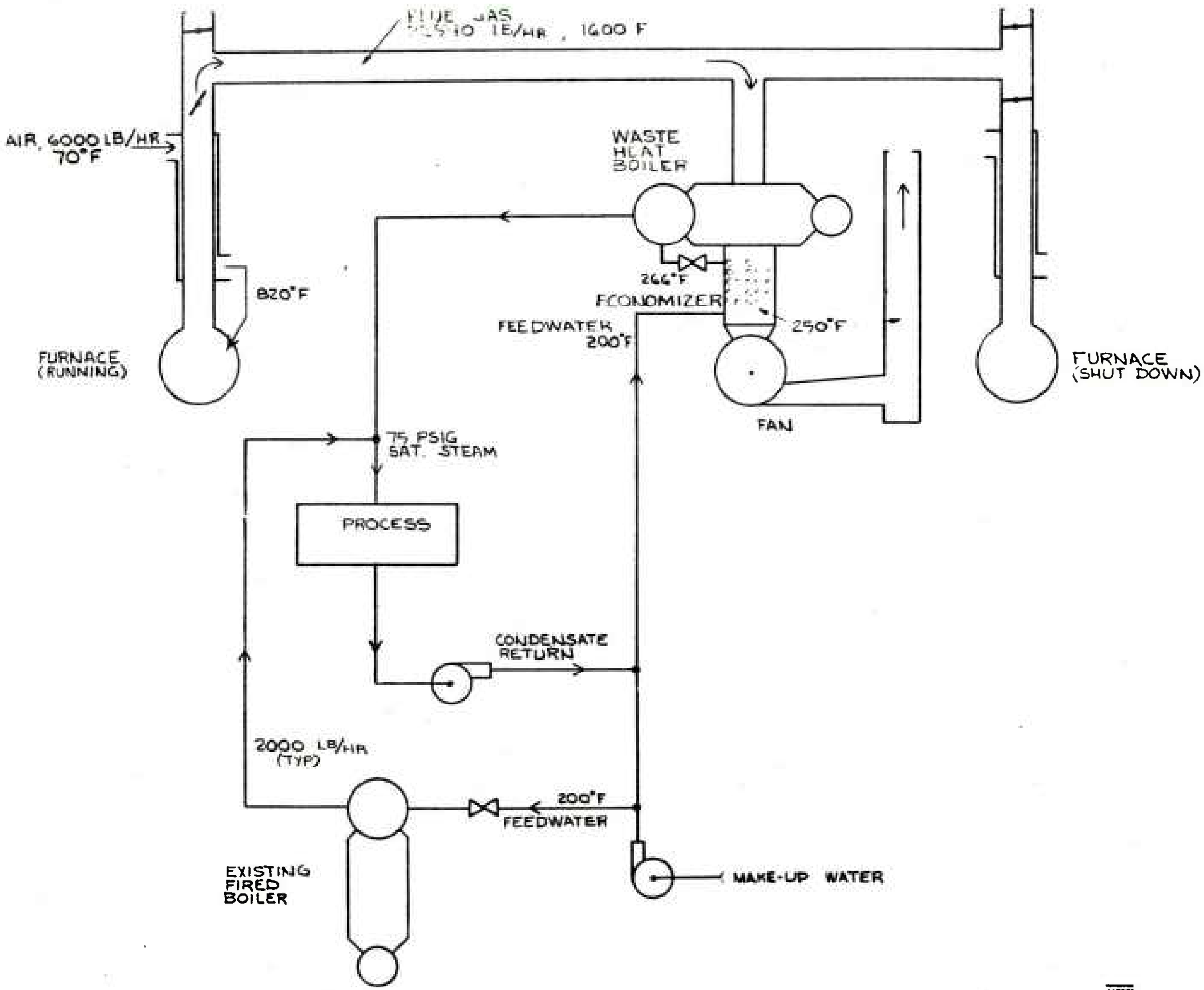

Figure 7. Flow schematic

802009

First, consider summer steam use. Based on an average of four representative work weeks (1979), the average steam use for a work day is 66,814 kg (147,325 lb). Based on a Saturday average, the steam use on a nonwork day is 38,291 kg (84,431 lb).

The rate of steam use during the nonworking hours of a week day is assumed the same as the Saturday rate. The rate of steam use during working hours is, therefore, 4448 kg/h (9807 lb/hr). Average steam use during nonworking hours is 1595 kg/h (3518 lb/hr).

Steam generated by a waste heat boiler operating at an exhaust temperature of 179°C (355°F) is as follows:

3670 kg/h (8092 lb/hr) while running

3337 kg/h (7357 lb/hr) while running

268 kg/h (590 lb/hr) additionally, if an economizer is added to bring the exhaust to 121°C (250°F).

These figures are based on the measured flows and temperatures, allowing for the recuperator, and for the duct losses.

The hourly steam schedule is shown in figure 8. The steam-use-day is assumed at 10 hours and the run mode of steam production is assumed at 17 hours. That is, the forging furnace runs billets for two shifts, producing stored heat for nearly an hour.

From the 1979 steam-use records, the heating season tapers off in April and begins again in October. Using a six-month heating season as an

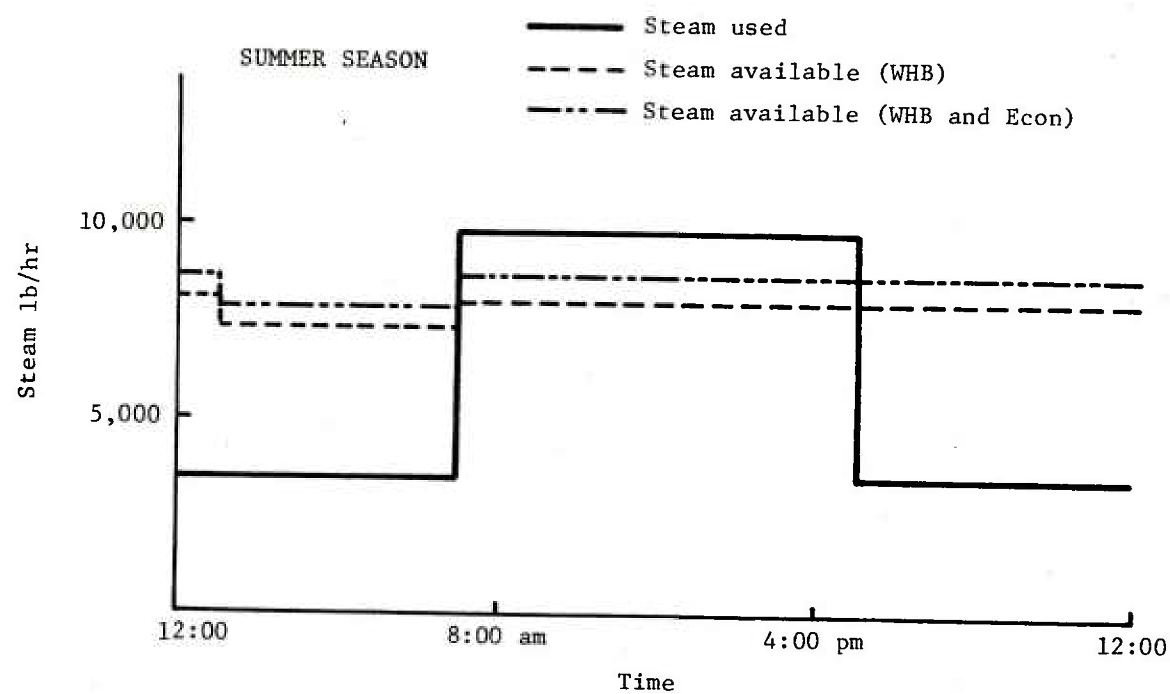

Figure 8. Steam utilization

801994

average, the hourly use rarely falls below waste heat boiler maximum production. It is assumed that all the steam generated during the heating season can be used.

The use of waste heat boiler steam is as follows:

Summer without economizer	$= 8.04 \times 10^6$ kg (17.73×10^6 lb)
Summer with economizer	$= 8.36 \times 10^6$ kg (18.44×10^6 lb)
Winter without economizer	$= 12.21 \times 10^6$ kg (26.92×10^6 lb)
Winter with economizer	$= 12.98 \times 10^6$ kg (28.62×10^6 lb)
Annually without economizer	$= 20.25 \times 10^6$ kg (44.65×10^6 lb)
Annually with economizer	$= 21.34 \times 10^6$ kg (47.06×10^6 lb)

The above calculations are based on concept 3, and assume no mobilization production rates.

ECONOMIC ANALYSIS - CONCEPT 3

Cost of Steam

Criteria to establish the cost of steam were based on the following factors:

- HHV 9217 kcal/m^3 $\left(1025 \text{ Btu/ft}^3\right)$ (per gas company)
- Boiler efficiency of 82% (per ARRADCOM)
- \$0.092/m^3 $\left(\$2.60/1000 \text{ ft}^3\right)$ (per ARRADCOM)
- 575.3 kcal/kg (1023.2 Btu/lb) steam (based on 5% blowdown and 93°C (200°F) feedwater)

The application of these factors resulted in the following cost equation:

$$\text{Cost} = \frac{1023.2\ \text{Btu}}{(\text{lb})\ \text{steam}} \times \frac{1}{0.82} \times \frac{\$2.60}{1000\ \text{ft}^3} \times \frac{\text{ft3}}{1025\ \text{Btu}}$$

$$= \frac{\$6.97}{1000\ \text{kg steam}} \left(\frac{\$3.16}{1000\ \text{lb steam}}\right)$$

Annual Savings in Gas Costs

Using the above cost equation, the annual savings in gas costs* are:

$141,300 without economizer

$148,700 with economizer

Annual Costs

Electricity, maintenance and operating expenses were established as follows:

Electric Power:	
Fan Power	$ 966
Miscellaneous	$ 34
Maintenance (average for first 10 years):	
5 man-days/yr at $25/hr	$1000
Materials	$ 500
Operating (estimated additional boiler watch time:	
240 hr/yr at $25/hr	$6000
	$8500

*Figures are rounded off to the nearest $100.

Net Annual Savings

By subtracting the annual operating costs ($8500) from the annual savings in gas costs, the net annual savings for concept 3 are:

$ 132,800 without economizer

$ 140,200 with economizer

First Costs

Calculations to establish the first costs for concept 3 were generated through manufacturers' estimates, vendor quotes and engineering estimates. (Refer to appendix B for an itemized list of suppliers and component weights.) First costs are as follows:

Description	Without Economizer ($)	With Economizer($)
Boiler, trim, F.W. regulator, nonreturn valve	51,000	63,000
Ducts, fabrication, insulation	17,400	17,400
Stack dampers, controls	10,700	10,700
Inlet plenum	5,200	5,200
Stack	1,000	1,000
Fan, motor, damper, controls	3,170	2,955
Expansion joints	3,000	3,000
Remote indicators	1,500	1,500
Structural steel	8,000	8,000
System erection	14,300	14,300
Project engineering	41,900	41,900
Steam piping	3,760	3,760
Shipping	2,340	2,340
Cold weather protection	7,500	7,500
Miscellaneous (startup assistance, debugging, ladders, and contingency)	15,000	15,500
	$185,770	$198,055

Payback Period

There are several different indicators of the economic value of a system such as this. The simplest measure is the payback period, which is defined as:

$$\frac{\text{First costs}}{\text{Net annual savings}} = \text{Payback period}$$

Payback periods, then, are as follows:

Without economizer: $\frac{\$185,800}{\$132,800} = 1.40$ years

With Economizer: $\frac{\$198,100}{\$140,200} = 1.41$ years

Economizer Alone: $\frac{\$12,285}{\$7,400} = 1.67$ years

Note that while the economizer increases the payback period of the overall system, it still has an attractive payback period when considered alone.

Return on Investment

More information regarding the value of the investment was obtained through the Return on Investment (ROI), which is defined as:

$$\text{ROI} = \frac{\text{Annual savings}}{\text{First costs}} \times 100$$

Boiler only:

Annual savings	=	$141,300
Annual costs	=	$8,500
Depreciation *	=	$18,580
First costs	=	$185,800

* Based on a conservative 10-year life and the straight-line method.

$$\text{ROI} = \frac{\$141,300 - \$8,500 - \$18,580}{\$185,800} = 61.5\%$$

Boiler with economizer:

Annual savings	=	$148,700
Annual costs	=	$8,500
Depreciation *	=	$19,810
First costs	=	$198,100

$$\text{ROI} = \frac{\$148,700 - \$8,500 - \$19,810}{\$198,100} = 61.8\%$$

Economizer:

Annual savings	=	$7,400
Annual costs	=	0
Depreciation *	=	$1,228
First costs	=	$12,285

$$\text{ROI} = \frac{\$7,400 - \$1,228}{\$12,285} = 50.1\%$$

Present Worth Analysis - Concept 3

A present worth analysis takes into account the lifetime costs, as well as the time value of money. For Case I, assume a 10-year life and 10% discount rate (i).

Boiler only:

First costs = $185,800

$$P \text{ (fuel)} = \$132,800 \times \frac{(1 + i)^{10} - 1}{i(1 + i)^{10}} = \$816,000$$

Present worth = $816,000 - $185,800 = $630,200

*Based on conservative 10-year life and straight-line method.

Economizer only:

First costs = \$12,285

$$P\ (\text{fuel}) = \$7400 \times \frac{(1+i)^{10}-1}{i\,(1+i)^{10}} = \$45{,}500$$

Present worth = \$45,500 - \$12,285 = \$32,215

For Case II, assume a 20-year life, 10% discount rate (i), and a retubing after 12 years costing 60% of the boiler/economizer first costs (less trim).

Boiler only:

Retubing cost = 0.60 x \$42,060 = \$25,200

$$P\ (\text{retubing}) = \$25{,}200\ (1+i)^{-12} = \$8000$$

First costs = \$185,800

$$P\ (\text{fuel}) = \$132{,}800 \times \frac{(1+i)^{20}-1}{i(1+i)^{20}} = \$1{,}130{,}600$$

Present worth = \$1,130,600 - \$185,800 - \$8000 = \$936,800

Economizer only:

Retubing cost = 0.60 x \$12,000 = \$7200

$$P\ (\text{retubing}) = \$7200\ (1+i)^{-12} = \$2300$$

First costs = \$12,285

$$P\ (\text{fuel}) = \$7400 \times \frac{(i+1)^{20}-1}{i(1+i)^{20}} = \$63{,}000$$

Present worth = \$63,000 - \$12,285 - \$2300 = \$48,415

The results of the present worth analysis are plotted on figure 9. This figure indicates the need to keep exhaust temperatures in the 121°C (250°F) range.

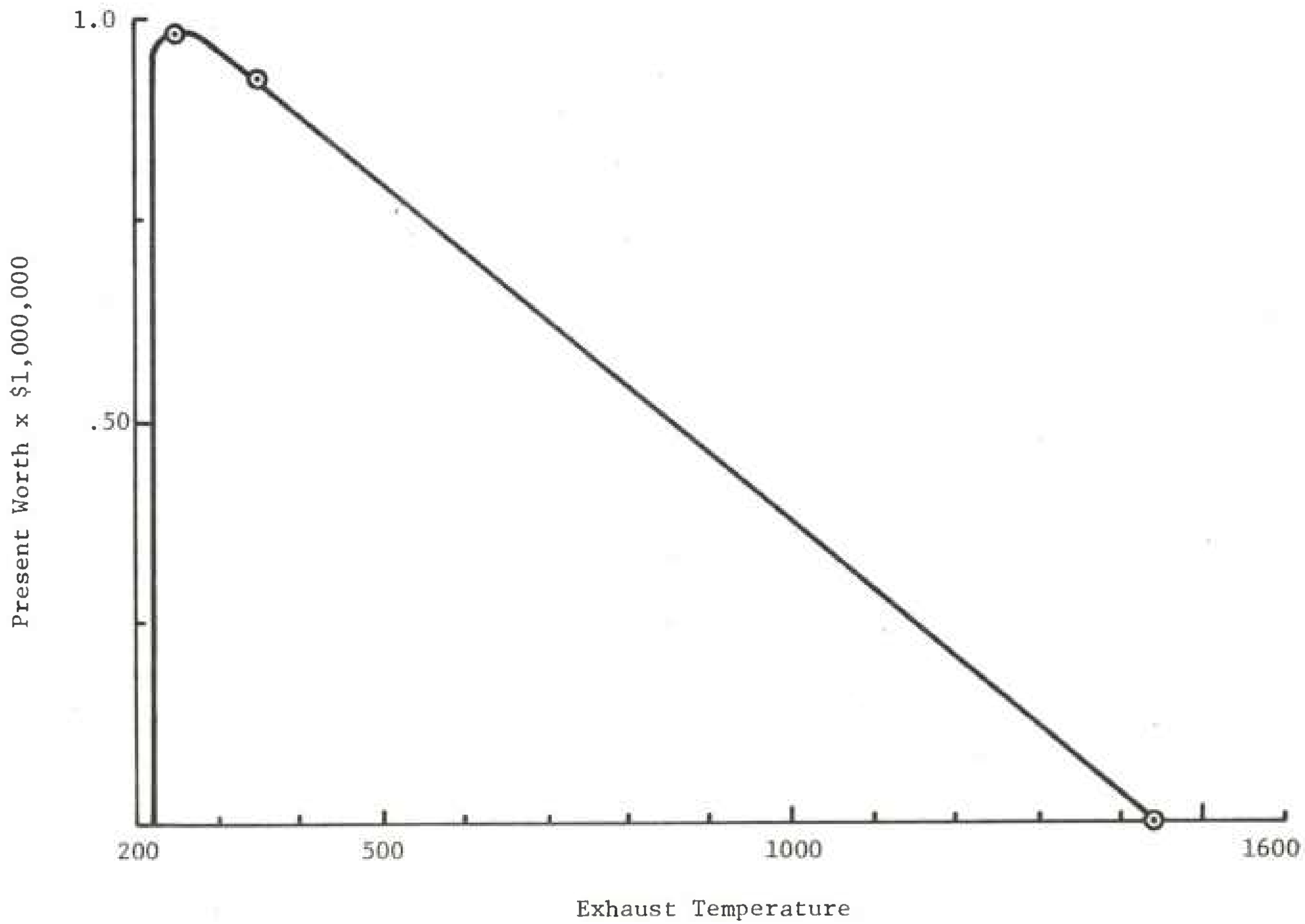

Figure 9. Present worth versus exhaust temperature (20 years)

801993

ECONOMIC ANALYSIS - CONCEPT 4

Annual Savings in Gas Costs

Criteria to establish the cost of steam were the same as for concept 3, excepting the exhaust temperature, which was based on 116°C (240°F); this represents a 1% increase in steam production over concept 3 due to flowing the exhaust of one furnace into a boiler sized for two furnaces.

Using the cost equation then, the annual savings in gas costs for concept 4 (including an economizer) are $149,800. Based on the same annual costs ($8500), the net annual savings are $141,300.

First Costs

Concept 4, as described, is sized for mobilization and includes an economizer. First costs are as follows:

Description	System with Economizer($)
Boiler, trim, F.W. regulator, nonreturn valve	70,000*
Ducting, insulation, fabrication	17,400
Dampers, controls	10,700
Inlet plenum	7,000*
Stack	1,400*
Fan, motor, controls	5,200*
Expansion joints	3,000
Remote indicators	1,500
Structural steel	12,000*
System erection	17,000*
Project engineering	41,900
Steam and feedwater piping	5,200*
Shipping	2,500*
Cold weather protection	8,000*
Miscellaneous - Start-up assistance, debugging, ladders, contingency	15,500
	$218,300

* Changed from concept 3 due to bigger boiler, fan.

Payback Period - No Mobilization

Earlier results have shown the economizer to be a cost-effective addition to the system. Those same results are evidenced in concept 4. The payback period is:

$$\frac{\$218{,}300 \text{ (First costs)}}{\$141{,}300 \text{ (Net annual savings)}} = 1.54 \text{ years}$$

Present Worth Analysis - No Mobilization

Because the preceding analysis assumes no mobilization rates, the present worth is less than the present worth of the concept 3 system. Concept 4, therefore, is next analyzed assuming an increased production rate, as well as utilization of waste heat from two stacks. While a mobilization lasting 20 years is perhaps not realistic, the following analysis is intended for comparison purposes only.

For this case, assume a 20-year life, a 10% discount rate (i), and a retubing after 12 years.

$$\text{First costs} = \$218{,}300$$

$$\text{Retubing cost} = 0.60 \times \$61{,}000 = \$36{,}600$$

$$\text{P (retubing)} = \$36{,}600\ (1 + i)^{-12} = \$11{,}700$$

$$\text{P (fuel)} = \$141{,}300 \times \frac{(1 + i)^{20} - 1}{i\ (1 + i)^{20}} = \$1{,}203{,}000$$

$$\text{Present worth} = \$1{,}203{,}000 - \$218{,}300 - \$36{,}600 = \$948{,}100$$

Payback Period - Mobilization

All first costs in this "mobilization scenario" are identical to concept 4. However, because the annual electricity costs and annual fuel savings double, the net annual fuel savings now becomes:

$$2 \times \$148,710 - 2 \times \$1000 - \$1500 - \$6000 = \$287,920$$

The payback period, therefore, is:

$$\frac{\$218,300 \quad \text{(First costs)}}{\$287,920 \quad \text{(Net annual savings)}} = 0.76 \text{ year}$$

Present Worth Analysis - Mobilization

For this case, again assume a 20-year life, a 10% discount rate (i), and a retubing after 12 years.

$$\text{First costs} = \$218,300$$

$$\text{Retubing cost} = 0.60 \times \$61,000 = \$36,600$$

$$\text{P (retubing)} = \$36,600\ (1 + i)^{-12} = \$11,700$$

$$\text{P (fuel)} = \$287,920 \times \frac{(i + 1)^{20} - 1}{i(1 + i)^{20}} = \$2,451,000$$

$$\text{Present worth} = \$2,451,000 - \$218,300 - \$36,600 = \$2,196,000$$

CONCLUSIONS

Waste heat recovery systems that retain the present stack and control arrangements are preferred for two reasons:

1. The back pressure control system is delicate but functional. There is, then, concern among operating personnel that a change in the system will cause serious control problems.
2. If, for any reason, the waste heat recovery system is down, production will continue as usual.

Concepts 3 and 4 satisfy the above requirements; both have ducts running from the tops of two stacks to a single waste heat boiler and fan. While either would be an excellent investment, concept 4 is selected because it is designed with a boiler and fan large enough to recover heat from both stacks at once; hence, it offers greater flexibility than a one-stack-at-a-time system (concept 3).

Additionally, concept 4 offers another important advantage in that it has the capability of generating waste heat steam during mobilization production; this is extremely important in relation to the activities at SAAP.

The economic comparisons in table 3 show that the three most likely systems (concept 3 without economizer, concept 4 without economizer, and concept 4 with economizer, at current production) have nearly the same present worth, as well as similar payback periods. Concept 4, including an

Table 3. Economic comparisons

	Concept 3 Without Economizer	Concept 3 With Economizer	Concept 4 With Economizer (No Mobilization)	Concept 4 With Economizer (Mobilization)
Payback Period (years)	1.40	1.41	1.54	0.76
Present Worth ($)	936,800	985,000	948,100	2,196,000

economizer and at mobilization production, offers a very quick payback period, even through its present worth is much greater than the other three systems. Further, given even a small possibility of mobilization production, concept 4 is again the best choice.

RECOMMENDATIONS

As a means of stabilizing the system and regulating the dilution air control valve, MTI suggests the addition of a variable inlet vane damper to the fan. The cost of this system adaptation would be quickly recovered by a reduction in fan power consumption. If the waste heat recovery system was added to the present system without correcting this hunting control, it would generate approximately three times more capacity than is required.

Recovering heat from the loading doors offers little incentive and, therefore, is not suggested. However, if it is desirable to exhaust the escaping flow for purposes of clean air, MTI recommends ducting to the waste heat boiler in order to minimize the heat loss. Such a system must be carefully designed to ensure the highest level of system performance.

Based on the comparison of concepts, in conjunction with the economic assessment, a waste heat recovery system representative of concept 4 (including an economizer) offers the greatest degree of flexibility and economy. Installation of such a system, sized for mobilization, is recommended as soon as possible.

System Control

The control concept is illustrated in figure 10 (all instrumentation is not shown). The controls will try to maintain 85 psig steam from the waste heat boiler. If the waste heat boiler is meeting the demand, the fired boiler will sense that its set point of 75 psig is exceeded and, therefore, will not generate steam. When the waste heat boiler fails to meet the steam demand, the fired boiler will cut in as steam pressure falls below 75 psig. Manual control will be used to start up and shut down the system.

Safety Control

Protection of plant personnel and protection of equipment are primary considerations in the design of a waste heat recovery system. Both of these considerations are taken into account as follows:

The steam side of the system should be designed according to ASME Section I recommendations. Boilers designed to this code have demonstrated remarkably good safety records; hence, there is little to be gained by going beyond code requirements.

Specifications call for personnel protection on high-temperature surfaces; the project engineer must make certain that the supplier complies with these requirements. Because it is likely that there will be small hot spots not anticipated by the vendor, the project engineer and plant personnel will have to handle minor deviations as they appear.

The waste heat recovery system should have as little effect as possible on the existing equipment, and failure in the system should not endanger the forge furnace system. Steps to be taken are as follows:

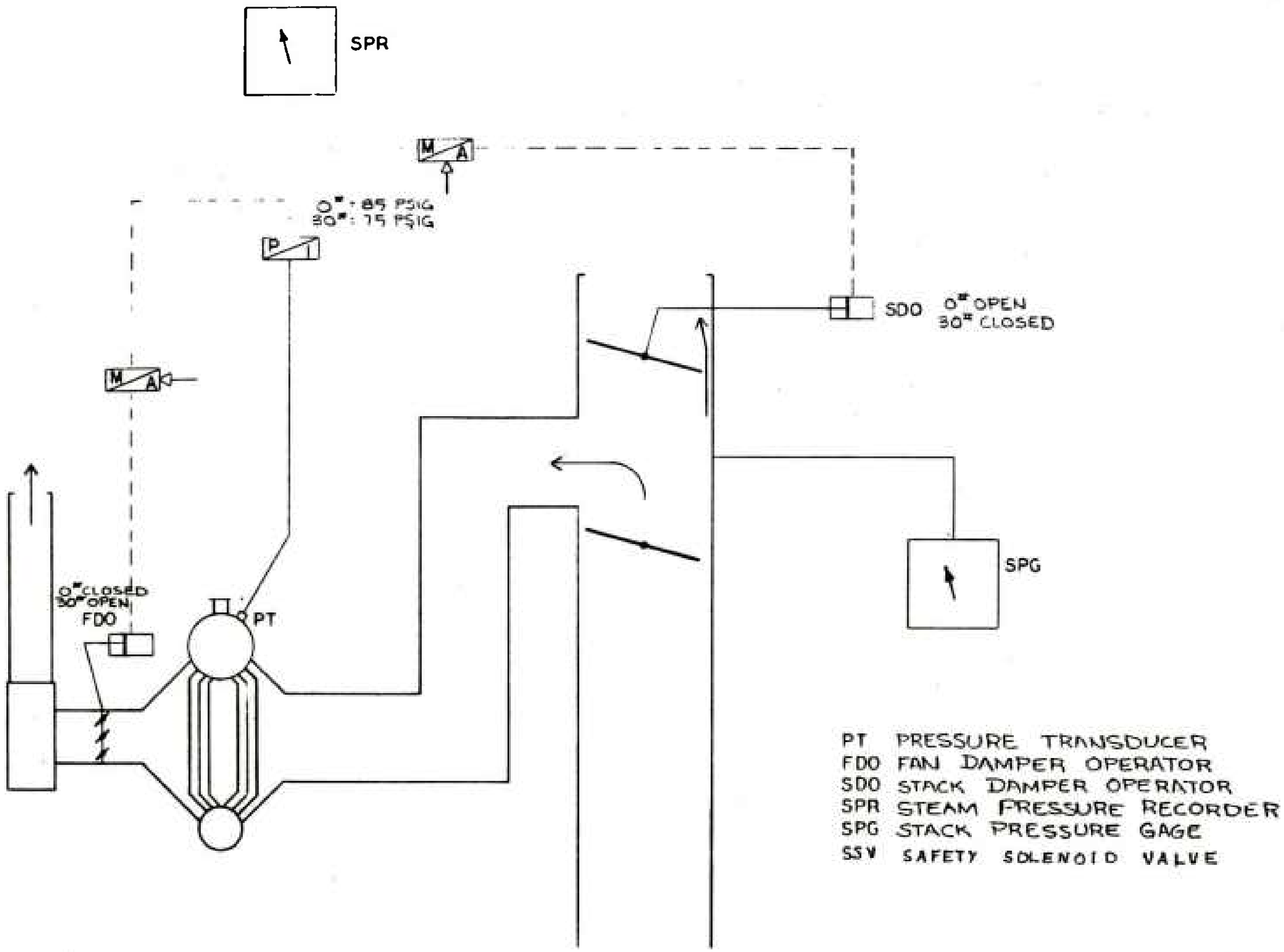

Figure 10. Control schematic

802015

1. In the event of a control air failure, the fan damper fails closed and the stack damper fails open.
2. If any excesses in stack pressure are detected, either of the dampers can be controlled manually.
3. The stack damper cannot be closed unless there is power to the fan. If the fan should lose power, the stack damper opens.
4. The boiler is equipped with high- and low-water level alarms as well as with a remote level indicator.

APPENDIX A. SAMPLE CALCULATIONS

Local Velocity

The temperature - density behavior of the combustion products will be closely approximated by the ideal gas equation:

$$\rho = \frac{P\ M}{RT} \tag{1}$$

P = pressure 14.7 $lb_f/in.^2$ or 2116.8 lb_f/ft^2

ρ = density (lb_m/ft^3)

M = molecular wt 29 lb/mole

R = gas constant 1545 ft $lb_f/°R$ mole

T = temperature (°R)

Solving for velocity from Bernoulli's equation:

$$(V_2^2 - V_1^2) = 2\,\frac{P_1 - P_2}{\rho} \tag{2}$$

where

V = velocity (ft/sec)

Referring to Figure A-1 and letting the subscript 1 indicate conditions at the inlet to the pitot tube where $V_1 = 0$, equation 2 becomes:

$$V_2 = \sqrt{2\frac{(P_1 - P_2)}{\rho}} \tag{3}$$

At a typical test point:

$P_1 - P_2 = 0.075$ in. W.C. $= 0.390\ lb_f/ft^2$

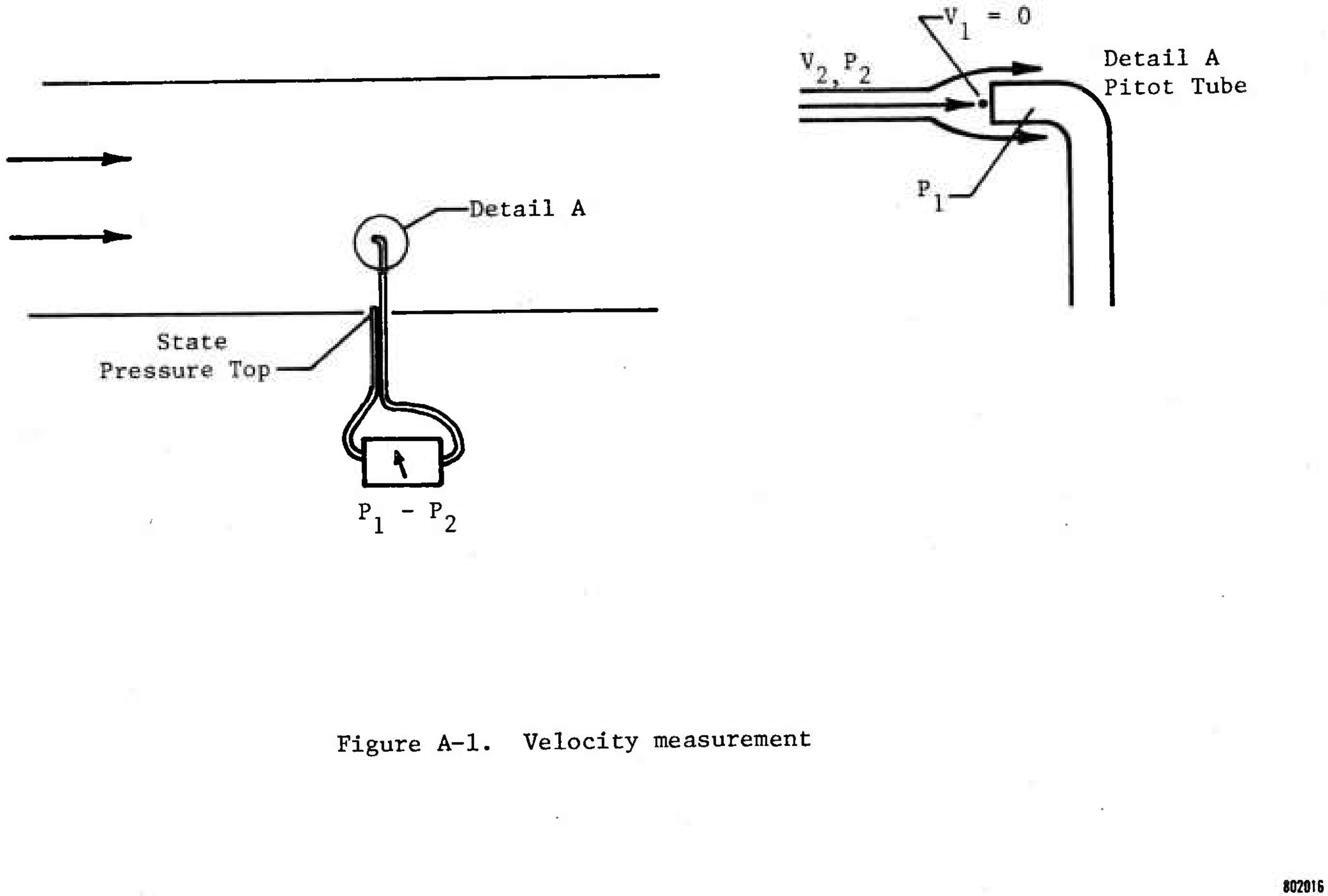

Figure A-1. Velocity measurement

from equation 1:

$$\rho = \frac{2116.8 \text{ lb}_f \ 29 \text{ lb}_m \text{ mole}°R}{ft^2 \text{ mole } 1545 \text{ ft/lb}_f \ (1755 + 460)°R}$$

$$\rho = 0.0179 \text{ lb}_m/ft^3$$

and

$$V_2 = \sqrt{\frac{2(0.390 \text{ lb}_f) \ ft^3}{ft^2 \ 0.0179 \text{ lb}_m}} \tag{4}$$

where

$$1 \text{ lb}_f = \frac{32.2 \text{ lb}_m \text{ ft}}{sec^2} \tag{5}$$

Substituting equation 5 into equation 4, the result is

$$V_2 = \sqrt{\frac{2 \ (0.390) \ ft^3 \ 32.2 \text{ lb}_m \text{ ft}}{sec^2 \ ft^2 \ 0.0179 \text{ lb}_m}}$$

or

$$V_2 = \sqrt{1403} \text{ ft/sec}$$

$$V_2 = 37.5 \text{ ft/sec}$$

<u>Discharging Door Radiation</u>

The radiation heat transfer equation is:

$$Q = FA\varepsilon\sigma \ (T_1^4 - T_2^4) \tag{6}$$

where:

Q = heat transfer (Btu/hr)

F = shape factor (dimensionless)

A = surface area (ft^2)

ε = surface emissivity (dimensionless)

σ = radiation constant 0.1714×10^{-8} Btu/hr ft^2 $°R^4$

T_1 = furnace interior temperature 2150°F or 2610°R

T_2 = surrounding temperature 90°F or 550°R

The shape factor from a relatively small opening to a large surrounding area is 1.0. The emissivity of an opening is nearly 1.0. A value of 0.9 will be assumed. The door is 20 in. x 24 in. or 3.33 ft^2.

Substituting the above numbers in equation 6, the result is

Q = 238,000 Btu/hr

APPENDIX B. SUPPLIER LIST

Item	Cost Estimate by	Alternate source (s) of supply
Boiler, economizer	Deltak Corp. P.O. Box 9496 Minneapolis, Minn. 55440	Henry Vogt Machine Co. 1000 W. Ormsby Ave. Louisville, Ky. 40210
Ducts, stack	Troy Boiler 2800 7th Ave. Troy, N.Y. 12180	A Local Fabricator or the Boiler Vendor
Duct insulation	A.P. Green Refractories Mexico, Mo. 65265	Carborundum* P.O. Box 490 Concordville, Pa. 19331
Stack dampers and controls	Frisch Dampers* Octapus Equipment Co. Buffalo, N.Y. 14221	Air Clean Dampers DaValco, Inc. 80 Main St. So. Bound Brook, N.J. 08880
Inlet plenum	MTI	The Boiler Vendor
Expansion joints	Troy Belting 2800 7th Ave Troy, N.Y. 12180	The Duct Fabricator
Fan, motor, controls	Buffalo Forge* 966 Broadway Albany, N.Y. 12204	Barry Blower 99 N.E. 77th Way Minneapolis, Minn. 55432 New York Blower* R.J. Wondrack Co. 700 E. Genesee St. Fayetteville, N.Y. 13066
Remote indicators	MTI	The Boiler Vendor
Structural steel	MTI	The Boiler Vendor or a Local Fabricator
System erection	MTI	A Local Construction Contractor
Project engineering	MTI	The Boiler Vendor or an Architect-Engineer
Steam and feedwater piping	Flach's Power Piping Glenmont, N.Y.	A Local Piping Contractor

*Manufacturer's representative.

Component Weights

Item	Estimated Weight kg (lb)	Estimated weight (operating) kg (lb)
Boiler, economizer	12,698 (28,000)	14,058 (31,000)
Ducts, stack	10,567 (23,300) total	10,567 (23,300) total
Dampers	860 (1900) each	860 (1900) each
Fan, motor	363 (800)	363 (800)
Structural steel	9100 (20,000)	9100 (20,000)

DISTRIBUTION LIST

Commander
US Army Armament Research & Development Command
ATTN: DRDAR-LCU-M (15)
DRDAR-TSS (5)
DRDAR-PRW-B
Dover, N.J. 07801

Administrator
Defense Technical Information Center
ATTN: Accessions Division (12)
Cameron Station
Alexandria, Va. 22314

Commander
DCASMA, Hartford (2)
96 Murphy Road
Hartford, Ct. 06114

Weapon System Concept Team/CSL
ATTN: DRDAR-ACW
Aberdeen Proving Ground, Md. 21010

Technical Library
ATTN: DRDAR-CLJ-L
Aberdeen Proving Ground, Md. 21010

Director
US Army Ballistic Research Laboratory
ARRADCOM
ATTN: DRDAR-TSB-S
Aberdeen Proving Ground, Md. 21005

Bener Weapons Laboratory
Technical Library
ATTN: DRDAR-LCD-TL
Watervliet, N.Y. 12189

Commander
US Army Armament Material Readiness Command
ATTN: DRSAR-LEP-L
Rock Island, Ill. 61299

US Army Material Systems Analysis Activity
ATTN: DRXSY-MP
Aberdeen Proving Ground, Md. 21005

Commander
US Army Munitions Production Base Modernization Agency
ATTN: SARPM-PBM-EC (2)
Dover, N.J. 07801

Commander
Scranton Army Ammunition Plant (5)
Scranton, Pa. 18501

Commander
Louisiana Army Ammunition Plant
Shreveport, L.A. 71130